

Thermal Analysis on Pin-Fins With Hexagonal & Threaded Geometry In Natural and Forced Convection

CH.V. Lakshmi¹, A. Vikranth², R. Tharun naidu³, J. Gowtham⁴, A. Vishnu⁵

¹ (Associate Professor, Department of Mechanical Engineering, SITAM-GVP, Andhra Pradesh, India)

^{2,3,4,5} (Department of Mechanical Engineering, SITAM-GVP, Andhra Pradesh, India)

To Cite this Article

CH.V. Lakshmi, A. Vikranth, R. Tharun naidu, J. Gowtham and A. Vishnu, "Thermal Analysis on Pin-Fins With Hexagonal & Threaded Geometry In Natural And Forced Convection", Journal of Science and Technology, Vol. 05, Issue 05, Sep-October 2020, pp16-27

Article Info

Received: 30-04-2020

Revised: 28-07-2020

Accepted: 30-07-2020

Published: 03-08-2020

Abstract: In the present work of heat transfer for hexagonal fins (1mm & 2mm) grooves on surface and threaded fin is addressed. The test has been performed on three different fin geometries having hexagonal (1mm)groove, hexagonal(2mm)groove, threaded fin(0.5mm)pitch and test performed by using a centrifugal blower, test section, heater and test panel and Results are obtained for temperature distribution, effectiveness, efficiencies at a same flow rate of air as it was conducted in forced convection and the same parameters considered for different values are obtained for natural convection with different fins as well. In this experiment for forced convection, the airflow rate is constant i.e, 2.3371 m/sec throughout the experiment. In natural convection, efficiency for the threaded fin is high with 93.89% and effectiveness of hexagonal(2mm)depth fin is 28.11. In forced convection, the efficiency of the threaded fin is high with 81.83% and effectiveness of hexagonal(1mm)depth fin is high with 23.51 was recorded. The heat transfer rate is higher in natural convection is hexagonal(2mm)depth fin with 11.41 watts and 21.75 watts in forced convection with hexagonal(1mm)depth fin.

Keywords: Pin-fins, Hexagonal fin, Threaded fin, Efficiency, Effectiveness.

I. Introduction

Convective Heat Transfer:

The process of heat transfer between the surface and surrounding fluid is known as convective heat transfer. In this process, the flow of energy is primarily due to the moment of fluid molecules.

They are two types of convective heat transfer:

1. Forced convection
2. Natural convection

Forced convection: In this type, the molecules of fluid are forced to move over the surface with the help of some external force. We can get a higher rate of heat transfer.

Natural convection: In this type, the fluid motion is caused by buoyancy forces that result from the density variations due to variations of temperature in the fluid.

Pin Fin:

Extended surfaces (fins) are frequently used in heat exchanging devices for the purpose of increasing the heat transfer between a primary surface and the surrounding fluid. Various types of fins, ranging from relatively simple shapes such as rectangular, square, cylindrical, annular, tapered or pin fins, to a combination of different geometries has been used.

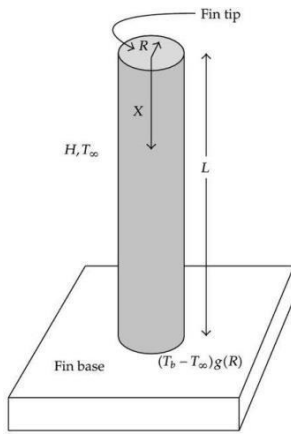


Fig a: Cylindrical fin

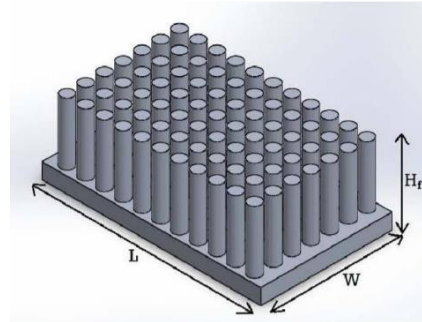


Fig b: Heat Sink

Dimensionless numbers:

1. **Prandtl number:** Is the ratio of kinematic viscosity to thermal diffusivity.

$P_r = \text{kinematic viscosity} / \text{thermal diffusivity}$

$$P_r = (\mu c_p) / K$$

$$P_r = (\mu / \rho) / (K / \rho \cdot C_p)$$

$$= \nu / \alpha$$

2. **Reynold's Number:** It may be physically interpreted as the ratio of inertia of force to viscous force in the velocity boundary layer. Large values of Re denotes high viscous forces.

$Re = \text{Inertia of force} / \text{viscous force}$

$$Re = \rho V D / \mu$$

3. **Grashof's number:** Grashof number (G_r) is a dimensional number used in heat transfer studies involving free or natural convection.

$$G_r = (\beta \Delta T D^3) / \nu^2$$

$$= (g \beta d^3 \Delta T) / \nu^2$$

$$= (\text{Inertia force} \times \text{Buoyancy force}) / (\text{viscous force})^2$$

4. **Nusselt number:** It is the ratio of convection heat transfer rate to the conduction heat transfer rate. Consider an internal flow in a channel of length L and the temperature at the lower and upper surfaces are T1 and T2 respectively.

The ratio is,

$$Nu = Q_{\text{conv}} / Q_{\text{cond}}$$

$$= [h A_c (T_1 - T_2)] / [(k A_c / L) (T_1 - T_2)]$$

$$= (h L) / k$$

II. Materials And Methods

Fin parameters:

Material - Aluminium alloy

Thermal conductivity (k) - 167 w/mk

For Hexagonal fin

$L = 15.7\text{cm}$

$D = 2.5\text{cm}$, $S = 1.2\text{cm}$

Depth-1mm, 2mm



Fig a: Cylindrical fin

For Threaded fin

$L = 15.7\text{cm}$

$D = 2.5\text{cm}$

Pitch = 0.5mm



Fig b: Heat Sink

Fins are fabricated in CNC machines and K-type thermocouple are used to measure temperature. Total fin length is 23.7cm, the thermocouple is placed on fins at a certain distance.

a. Experimental set-up:

Methodology: A pin fin with different geometry (Hexagon, Threaded) cross-section is fitted across a long rectangular duct. The other end of the duct is connected to the suction side of a blower and the air flows pass through the fin perpendicular to the axis. One end of the projects outside of the duct and is heated by a heater. The temperature at 5 points along the length of the fin are measured by K-type thermocouples connected along the length of fin the area flow rate is measured by an orifice meter fitted on the delivery side of the blower. This is for forced convection. For natural convection, it removes the duct and off the blower connects the fin with the heater and notes the temperature readings.

b. Specifications of apparatus:

- 1 Duct size = $150 \times 100\text{mm}$
2. Diameter of orifice = 20mm
3. Diameter of delivery pipe = 50mm
- 4 Number of thermocouples on fin = 5
- 5 Thermocouple (6) reads ambient temperature inside the duct
6. Temperature indicator = $0-300^\circ$
- 7 Dimmer stat for heat input control = 230v, 2Amp
- 8 Fluid used in manometer-Mercury (Density- 13600kg/m^3)



a. Forced convection set-up



b. Natural convection set-up

Cross Section Area Values:

For hexagonal fin(1mm),

$$A_c = 3.711 \times 10^{-4} \text{ m}^2$$

For hexagonal fin(2mm),

$$A_c = 3.681 \times 10^{-4} \text{ m}^2$$

For threaded fin,

$$A_c = 4.9087 \times 10^{-4} \text{ m}^2$$

Surface Area Values:

For hexagonal fin(1mm),

$$A_s = 0.01204 \text{ m}^2$$

For hexagonal fin(2mm),

$$A_s = 0.01203 \text{ m}^2$$

For threaded fin,

$$A_s = 0.01233 \text{ m}^2$$

Perimeter:

For hexagonal fins,

$$P = 0.072 \text{ m}$$

For threaded fin,

$$P = 0.0785 \text{ m}$$

For, irregular shapes we use hydrodynamic diameter

$$D_h = 4A_c/P$$

For hexagonal fin,

For, 1mm

$$D_h = 0.02061 \text{ m}$$

For hexagonal fin,

For, 2mm

$D_h = 0.02045\text{m}$

In case of the threaded fin, D_h = diameter of fin.

$D_h = D_{\text{fin}}$

$D_h = 0.025\text{m}$

III. Results and Discussion

The models of hexagonal & threaded fins are tested under different voltages of the same diameter and same material. It is observed that heat transfer rates and temperatures have varied with varying geometries. By the depth of the groove increases the heat transfer rate also changes in hexagonal fins. By comparison of threaded and hexagonal two different depths of the groove, the heat transfer rate is higher in hexagonal fin. Here are the observations of different pin fins taken during experimentation

For Natural convection

Table 3.1: Observations during experimentation for Hexagon fin (1.0mm)

S.No	Power input (W)		Fin Temperatures($^{\circ}\text{C}$)					Duct Fluid Temperatures ($^{\circ}\text{C}$)
	V	A	T_1	T_2	T_3	T_4	T_5	
1	80	0.45	121.3	119.3	116.3	114.5	111.2	54.2
2	90	0.50	180.2	178.2	176.7	174.2	172.3	55.4

Table 3.2: Observation during experimentation for Hexagon fin (2.0mm)

S.No	Power input(W)		Fin Temperatures($^{\circ}\text{C}$)					Duct Fluid Temperatures ($^{\circ}\text{C}$)
	V	A	T_1	T_2	T_3	T_4	T_5	
1	80	0.45	130.2	123.7	120.3	118.5	117.5	57.1
2	90	0.50	187.9	179.2	175.0	170.6	171.0	58.1

Table 3.3: Observations during Experimentation for Threaded Fin

S.No	Power input (W)		Fin Temperatures (°C)					Duct Fluid Temperature (°C)
	V	A	T ₁	T ₂	T ₃	T ₄	T ₅	
1	80	0.45	100.1	101.2	92.3	96.7	94.0	53.5
2	90	0.50	143.1	130.2	117.3	126.7	122.4	55.6

It shows the values of surface temperatures (C°) of different fins i.e, hexagonal (1mm)depth, hexagonal (2mm)depth, threaded fin in natural convection. V is the voltage given as heat input. T₁, T₂, T₃, T₄, T₅ are the surface temperatures of the fin at different lengths. T₆ is the ambient temperature (or) temperature of the air surrounding to fin during experimentation.

For Forced convection

Table 3.4: Observations during experimentation for Hexagon fin (1.0mm)

S.No	Power input (W)		h(mm)	Fin Temperatures(°C)					Duct Fluid Temperatures (°C)
	V	A		T ₁	T ₂	T ₃	T ₄	T ₅	
1	80	0.45	8	119.0	116.4	111.8	109.4	106.5	58.0
2	90	0.45	8	148.1	144.7	139.1	136.1	131.5	60.5

Table 3.5: Observation during experimentation for Hexagon fin (2.0mm)

S.No	Power input(W)		h(m m)	Fin Temperatures(°C)					Duct Fluid Temperatures (°C)
	V	A		T ₁	T ₂	T ₃	T ₄	T ₅	
1	80	0.45	8	109.5	106.2	105.2	103.6	98.3	57.0
2	90	0.50	8	134.3	130.3	127.9	126.6	118.6	60.6

Table 3.6: Observations during Experimentation for Threaded Fin

S.No	Power input (W)		h(mm)	Fin Temperatures (°C)					uct Fluid Temperature (°C)
	V	A		T ₁	T ₂	T ₃	T ₄	T ₅	
1	80	0.45	8	90.5	91.8	96.1	98.8	108.9	57.8
2	90	0.50	8	109.1	110.3	115.7	119.2	131.8	60.2

It shows the values of surface temperatures(C°) of different fins i.e, hexagonal (1mm)depth, hexagonal (2mm)depth, threaded fin in forced convection. V is the voltage given as heat input. T₁, T₂, T₃, T₄, T₅ are the surface temperatures of the fin at different lengths. T₆ is the ambient temperature (or) temperature of the air surrounding to fin during experimentation. h(mm) is the manometric reading. Mercury is used as a manometric fluid.

Table 3.7: Values for dimensionless numbers in natural convection

V	β	Gr	Ra	Nu	h (w/m ² k)	m	Fin type
80	2.79×10^{-3}	3.35×10^4	2.32×10^4	6.543	9.673	3.352	Hexagon(1m m)
80	2.73×10^{-3}	3.045×10^4	2.10×10^4	6.381	9.76	3.381	Hexagon(2m m)
80	2.86×10^{-3}	4.961×10^4	3.44×10^4	7.21	8.56	2.86	Threaded
90	2.57×10^{-3}	4.991×10^4	3.434×10^4	7.216	11.23	3.613	Hexagon(1m m)
90	2.56×10^{-3}	4.762×10^4	3.27×10^4	7.13	11.19	3.620	Hexagon(2m m)
90	2.74×10^{-3}	6.22×10^4	4.29×10^4	7.628	9.59	3.023	Threaded

It shows the values of some dimensionless numbers in natural convection. V is the voltage given as heat input, β is the volumetric coefficient of thermal expansion, Gr is the grashof number, Ra is the rayleigh number, Nu is the nusselt number, h is the heat transfer coefficient, m is the fin property.

Table 3.8: Values for dimensionless numbers in forced convection

V	β	h (m)	Va (m/sec)	Re	Nu	h (w/m ² k)	m	Fin type
80	0.4	0.08	2.3371	2284.10	22.22	32.85	6.178	Hexagon(1 mm)
80	0.4	0.08	2.3371	2266.17	22.14	32.99	6.21	Hexagon(2 mm)
80	0.4	0.08	2.3371	2918.45	24.93	29.58	5.32	Threaded
90	0.4	0.08	2.3371	2082.47	21.24	33.09	6.20	Hexagon(1 mm)
90	0.4	0.08	2.3371	2162.61	21.64	33.10	6.22	Hexagon(2 mm)
90	0.4	0.08	2.3371	2770.38	24.31	29.63	5.35	Threaded

It shows the values of some dimensionless numbers in forced convection. V is the voltage given as heat input, β is the volumetric coefficient of thermal expansion, h is the manometric reading, Va is the velocity of air passes on the fin, Re is the Reynolds number, Nu is the nusselt number, h is the heat transfer coefficient, m is the fin property.

Table 3.9: Observation for Hexagon(1.0mm), Hexagon(2.0 mm), Threaded fin's efficiencies and effectiveness in natural convection

S. No	Voltage(V)	Surface Temp (C°)	η (%)	ϵ	Fin type
1	80	116.52	91.6	27.92	Hexagon(1.0mm)
2	80	122.04	91.56	28.11	Hexagon(2.0mm)
3	80	98.86	93.89	23.52	Threaded
4	90	176.32	90.50	27.56	Hexagon(1.0mm)
5	90	176.74	90.47	27.77	Hexagon(2.0mm)
6	90	127.94	93.11	23.38	Threaded

It shows the values of efficiencies and effectiveness of different fins in natural convection. V is the voltage given as heat input, surface (C°) is the surface temperature of a fin, η is the efficiency of the fin, ϵ is the effectiveness of fin.

Table 3.10: Observation for Hexagon(1.0mm), Hexagon(2.0 mm), Threaded fin's efficiencies and effectiveness in forced convection

S.No	Voltage(v)	Surface θ	η (%)	ϵ	Fin type
1	80	112.62	77.18	23.51	Hexagon(1.0mm)
2	80	104.56	76.98	23.29	Hexagon(2.0mm)
3	80	97.22	81.83	20.53	Threaded
4	90	139.94	77.07	23.47	Hexagon(1.0mm)
5	90	127.54	77.0	23.60	Hexagon(2.0mm)
6	90	117.22	81.73	20.58	Threaded

It shows the values of efficiencies and effectiveness of different fins in forced convection. V is the voltage given as heat input, surface ($^{\circ}\text{C}$) is the surface temperature of a fin, η is the efficiency of the fin, ϵ is the effectiveness of fin.

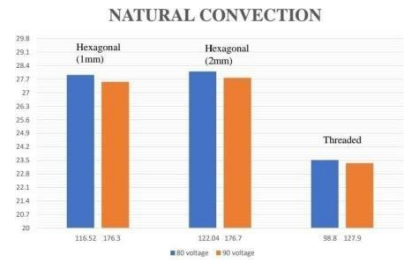
a. GRAPHS:

Efficiency vs surface temperature ($^{\circ}\text{C}$) -



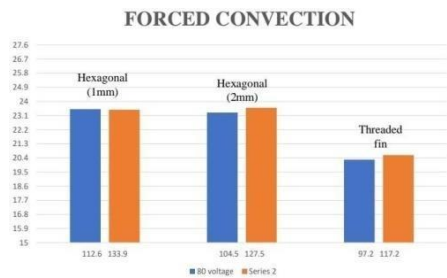
From the graph, it can be observed that in natural convection the surface temperature is low and the efficiency is high in all fins. For threaded fins, the efficiency is higher among other fins.

Effectiveness vs surface temperature (°c)-



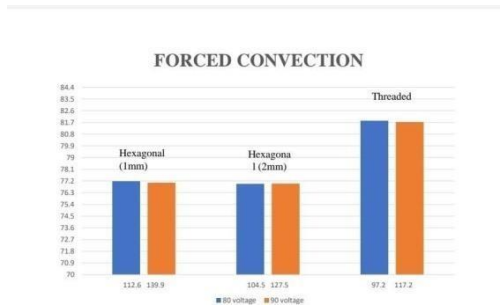
From the graph, it can be observed that in natural convection the surface temperature is low and the effectiveness is high in all fins. For, hexagonal fin(2mm) has high effectiveness among two other fins.

Effectiveness vs surface temperature (°c) -



From the graph, it can be observed that in forced convection the surface temperature is high and the effectiveness is high in all fins. For, hexagonal fin(2mm) has high effectiveness among two other fins.

Efficiency vs surface temperature (°c) -



From the graph, it can be observed that in forced convection the surface temperature is low, the efficiency is high but in hexagonal fin (2mm) the surface temperature is high and efficiency also high.

IV. Conclusion

Heat transfer rates depend not only on the heat transfer coefficient but also the surface area changes and different geometries. By changing the different geometries of fins, the heat transfer rates can be enhanced and are compared to those of the normal cylindrical fins at the same base temperatures. It is concluded that two types of convections natural and forced, give different heat transfer values.

For natural convection, the surface temperature is higher in hexagonal fin (2mm)depth. By comparing the efficiencies threaded fin has maximum and hexagonal fin(2mm)depth has a minimum. By comparing the effectiveness hexagonal fin(2mm)depth has maximum and threaded fin has a minimum. Hexagonal fin(2mm)depth has a high heat transfer rate among two fins.

For forced convection, the surface temperature is higher in hexagonal fin(1mm)depth. By comparing the efficiencies threaded fin has maximum and hexagonal fin (1mm)depth has a minimum. By comparing the effectiveness hexagonal fin (2mm)depth has maximum and threaded fin has a minimum. Hexagonal fin(2mm)depth has a high heat transfer rate in natural convection and hexagonal fin (1mm)depth has a high heat transfer rate in forced convection.

By comparing the experimental and theoretical calculations the percentage of error (%) is below 5 i.e, around 3.75.

V. Future Scope

In this study, only one design parameter is taken i.e, same diameter and same material. In forced convection, the airflow rate is constant 2.3371 m/sec in all types of fins. In future by changing the depths of the groove and by varying airflow rates and changing material it can also be performed.

Nomenclature:

- D = Diameter of pin fin (m)
- D_h = Hydrodynamic diameter of pin fin(m)
- A_s = Surface area of fin(m^2)
- A_c = Cross section area of fin(m^2)
- ν = Kinematic viscosity(m^2/sec)
- k = Thermal conductivity(w/mk)
- h = Heat transfer coefficient (w/ m^2k)
- p = Perimeter of fin (m)
- L = Length of pin (m)
- L_c = Characteristics length of fin(m)
- T_s = Surface temperature of fin ($^{\circ}C$)
- T_a = Ambient temperature of air ($^{\circ}C$)

- T_m = Mean temperature ($^{\circ}\text{C}$)
- C_d = Coefficient of discharge (0.64)
- h = Difference of level in manometer
- η = Efficiency of fin
- Σ = Effectiveness of fin

References

- [1] Hani A. El-Sheikh and Suresh v. Garimella, June 2000, Enhancement of air jet impingement Heat Transfer Using Pin-Fin Heat Sinks, IEEE Transactions on Components and Packaging Technology, VOL. 23, No.2.
- [2] Denpong Soodphakdee, MasudBehnia, and David Watabe Copeland, 2001, A Comparison of Fin Geometries for Heat sinks in Laminar Forced Convection: Part I – Round, Elliptical, and Plate Fins in Staggered and In-line Configurations. The International Journal of Microcircuits and Electronic Packaging, Volume 24, Number 1, First Quarter, (ISSN1063-1674).
- [3] O. N. Sara, S. Yapici, M Yilmaz, 2001, Second law analysis of rectangular channels with square pin-fins, Int. comm...Heat and mass transfer Vol 28. No. 5, pp 617-630.
- [4] O. N. Sara, 2003, Performance analysis of rectangular ducts with staggered square pin fins, Energy Conversion and Management 44:1787-803.
- [5] YoavPeles, Ali Kosar, Chandan Mishra, Chih-Jung Kuo, Brandon Schneider, 2005, Convective heat transfer across a pin fin micro heat sink, International Journal of Heat and Mass Transfer 48 3615-3627.S.
- [6] N. Sahiti, F. Durst, A. Dewan, 2005, Heat transfer enhancement by pin elements, International Journal of Heat and Mass Transfer 48 4738-4747.
- [7] Ali Kosar and YoavPeles, February 2006, Thermal-Hydraulic Performance of MEMS-based Pin Fin Heat sinks, Journal of Heat Transfer, ASME, Vol. 128/121.
- [8] PaisarnNaphon, AnusornSookhasem, 2007, Investigations on heat transfer characteristics of tapered cylinder pin fin heat sinks, 0196-8904/\$ - see front matter 2007 Published by Elsevier Ltd.doi: 10.1016/j.enconman.2007.04.020.
- [9] Pitchandi K, Natarajan E, December 2008, Entropy Generation in Pin Fins of Circular and Elliptical cross-sections in Forced convection with Air, Int. J. of Thermodynamics Vol. 11 (No. 4), pp.161-171.
- [10] Ahmad Khoshnevis, FaramarzTalat, MaziyaJalaal, Esmail; Esmailzadeh, May 2009, Heat Transfer Enhancement of Slot and Hole Shape Perforations In Rectangular Ribs of a 3-D Channel, 17th Annual (International) Conference on Mechanical Engineering –ISME2009, University of Tehran, Iran.
- [11] GongnanXie, BengtSunden, Lieke Wang and EsaUtriainen, August 2009, Augmented heat transfer of an internal blade tip by full or partial arrays of pin-fins, Int. Symp. on Heat Transfer in Gas Turbine Systems 9-14, Antalya, Turkey.
- [12] Abdullah H. Alyssa, Ayman M. Maqableh and ShathaAmmourah, October 2009, Enhancement of natural convection heat transfer from a fin by rectangular perforations with an aspect ratio of two, International Journal of Physical Sciences Vol. 4 (10), pp. 540-547.
- [13] Abdullah H. Alessa and Mohammad q. Al-Odat, 2009, Enhancement of natural convection heat transfer from a fin by triangular perforations of bases parallel and toward its base. The Arabian Journal for Science and Engineering, Volume 34, Number 2B.
- [14] Santosh Krishnamurthy, YoavPeles, Journal of Heat Transfer, APRIL 2010, Flow Boiling Heat Transfer on Micro Pin Fins Entrenched in a Microchannel, ASME, and Vol.132/041007-1.
- [15] S. H. Barhatte, M. R. Chopade, V. N. Kapatkar, Experimental and computational analysis and optimization for heat transfer through fins with different types of the notch, Journal of Engineering Research and Studies.
- [16] Er. R.K. Rajput, Heat and Mass Transfer, S.Chand and company Pvt. LTD.
- [17] J P Holman, Heat Transfer, McGraw Hill Publishing Company Limited.
- [18] C P Kothandaraman, S Subramanyam, Heat and Mass Transfer data book, New Age International Publishers.